Design and Development of Ball Bearing for Inclined Loads

by

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

Approved by,

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July 2010

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

(AHMAD RAFIE BIN MOHMAD TAHIR)

ABSTRACT

A journal bearing is a mechanical component that is used to allow constrained relative motion between two or more parts, typically rotation or linear movement. Bearings may be classified generally according to the motions they allow and according to their principle of operation as well as by the directions of applied loads they can handle. Typically, ball bearings are designed to accommodate either axial or radial load. They are not capable of handling both the load simultaneously. A slewing bearing is a special type of bearing that can accommodate axial as well as radial load simultaneously. At present, their application including heavy cranes, large wind turbine, excavation machinery and all kinds of harbor and shipyard cranes. For smaller loads, there is no slewing bearing that has been design due to the complexity of its manufacturing. This research study could be looking into the design aspects of slewing bearing and its manufacturing technology for small loads applications. This paper contains the fundamental understanding of slewing bearing design and its calculation theories. Also, this paper included the calculation results and also the detail design of the slewing bearing.

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CHAPTER 1

INTRODUCTION

1.1 Background of Study

The word ‘slewing’, by definition, is the rotation around an axis which is usually the z-axis. Normal journal bearings can only accommodate axial or radial load only whereas slewing bearings can accommodate axial, radial and moment loads acting either singly or in combination and in any direction[1]. A slewing bearing is basically a large-sized bearing which may also have gear teeth to receive the energy of a motor, in accordance with the desired application. They are chiefly employed in a horizontal position for transferring axial forces and large tilting moments. External loads such as axial force, radial force and moments load are transferred from the rotating structure through the bearing rings and rolling elements to the supporting structure (Figure 1.1). They are not mounted on a shaft and in housing. The rings, which are simply bolted on the seat surface are available in one of three executions;

1. Without gears
2. With an internal gear
3. With an external gear

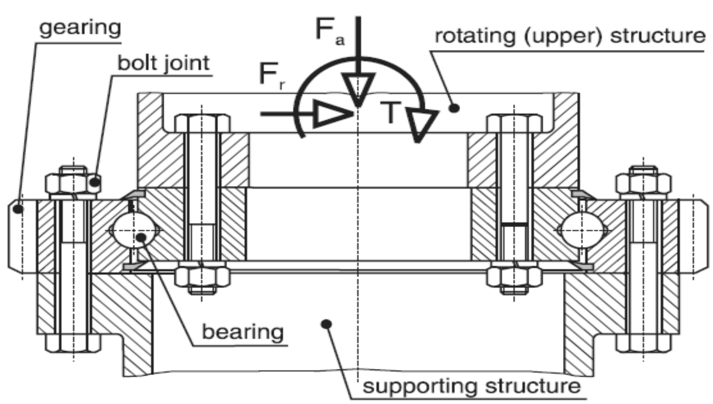


Figure 1.1 Elements of a rotational connection with a single row four contact ball bearing

Slewing bearing can be designed as ball bearings or roller bearings. The cage consists of a number of segments or separators which space out the rolling elements. This type of bearing is usually used for huge application and can vary from tower cranes to wind-power generators, excavation machinery, offshore equipments, such as offshore platforms, all kinds of harbor and shipyard cranes, and deck cranes[2].

1.2 Problem Statement

Usually, for small applications, slewing ring was not used as a bearing. For small applications usually a combination of thrust and journal ball bearing were used to support, let say, a shaft. Therefore, this project will aim to design a small slewing bearing for small application to sustain inclined loads. Example of machines that can use this type of bearing is small robotic arm, welding machine table, radar and many more.

In this project, a study on the design of a slewing ring will be conducted. From the knowledge gained, a small slewing bearing will be designed. After designing was completed, the design will be fabricated and tested.

1.3 Objective and Scope of Study

The main objectives of this research are:

1. To design a slewing bearing using CAE software.
2. To fabricate the slewing bearing using available material and method.
3. To test the slewing bearing for estimated inclined load of at least 10kg

CHAPTER 2

LITERATURE REVIEW AND/OR THEORY

From the review of several books on bearings, some journals and even website of companies that produce bearings, these information below was noted for its relevancy regarding this project;

2.1 Slewing Bearing in General

Slewing bearings (Figure 2.1) are large rolling bearings used for supporting excavators, cranes, and similar equipment in heavy machinery construction where mounting space is restricted, loads are high and operational reliability is important. The bearings can take up radial and axial forces as well as tilting moments. They are chiefly employed in a horizontal position for transferring axial forces, radial forces and large tilting moments. Radial forces are generally subsidiary. [3]

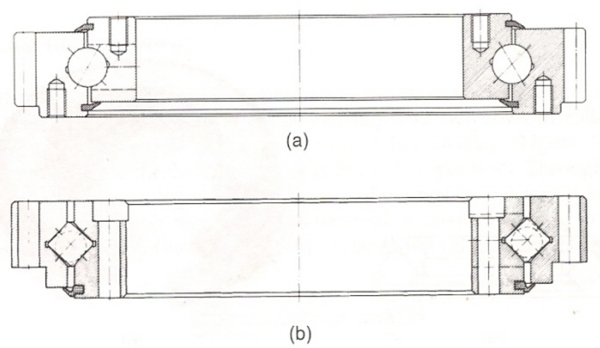


Figure 2.1 Slewing bearings: (a) four-point bearing (b) cross-roller bearing [3]

2.2 Constant Combined Load

In slewing bearing application, it is often related to the combination of loads which are the axial forces, radial forces and also the moments forces. In bearing engineering, the term combined load refers to cases where the bearing is stressed by a force *F* engaging at the so called load angle β (Figure 2.2). The term combined load indicates the replacement of a constant oblique load F by its components $F_r = F\cos\beta$ and $F_a = F\sin\beta$ that are used in the calculations. The equivalent dynamic load P on a bearing under combined load is calculated using the pressure distribution and fatigue theory. According to fatigue theory, the mean rolling element load

$$Q_m = \left(\frac{1}{z}\sum_{i=1}^{z} Q_i^p\right)^{\frac{1}{p}}$$

Is proportional to the dynamic stressing of a rolling bearing and represents the force value critical in the calculation. [3]

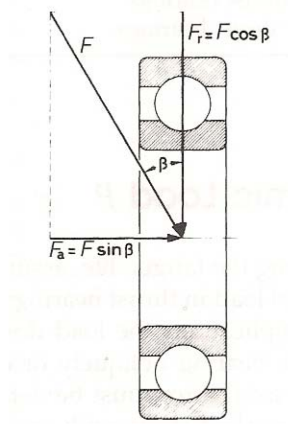


Figure 2.2 Combined loading of a radial deep-groove ball bearing [3]

2.3 Load Distribution in a Four Contact Point Slewing Bearing

2.3.1 Calculation of Load Distribution

Calculation of load distribution in a bearing is crucial to find the limiting contact pressure created between the rolling elements and the raceways [4]. From the load distribution also, the dimension of the slewing bearing can be determined. Besides that, it tells us what the maximum load is and indicates which rolling element carries the heaviest load. Load distribution also provides information for determining an equivalent load for calculating the dynamic capacity of the slewing bearing from the particular loads of each rolling element [4].

The procedure for determining the load distribution of slewing bearing provides information on the general rigidity of the bearing which is based on the relative displacements (axial, radial and rotational). In simplified form, rolling bearings may be considered as springs with axial and radial elasticity as well as resilience to tilting. The schematic diagram was shown in Figure 2.3 [3]. Under certain load F_r and moment M acting on it, the bearing is displaced radially by δ_r; under action of F_z, it is displaced by δ_z, the rotation angle is θ.

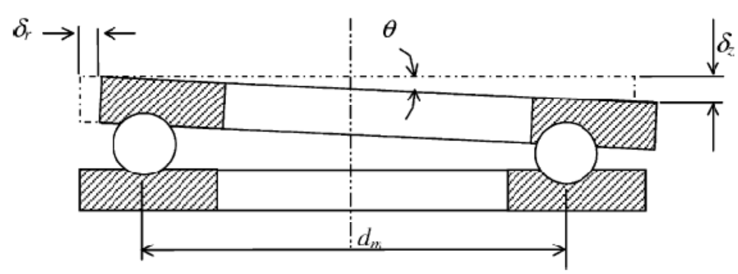


Figure 2.3 Relative displacements between the raceways [4]

For simplification purposes, the calculation procedure used supposes that the raceways are rigid and only assumes elastic deformations of the contacts of the rolling elements and raceways (Hertz contacts). Neither does it takes into account the effects caused by the supporting surface of the bearing [4].

2.3.2 Load Distribution Calculation Method

The ball bearings carry a different load in accordance with the position it occupies within the bearing and with the geometric characteristics of the raceway and the material properties. This is caused by the combined loads (axial, radial and moment).

The turning speed of these bearings is low enough for the effects of the gyroscopic forces and centrifugal forces of the ball bearings themselves to be disregarded.

As mentioned above in section 2.3.1, there are three types of relative displacements between the raceways in all loaded bearings:

- axial displacement δ_z,
- radial displacement δ_r,
- rotation θ.

These relative displacements define the position of the loci of the centres of curvature of the two raceways in reference to their initial positions, and they are shown in the following equation:

$$s = f(\delta z, \delta r, \theta, A) \tag{1}$$

where s is the relative distance between the centres of curvature of the raceways and A is the initial relative distance between the centres of curvature of the raceways (Figure 2.4).

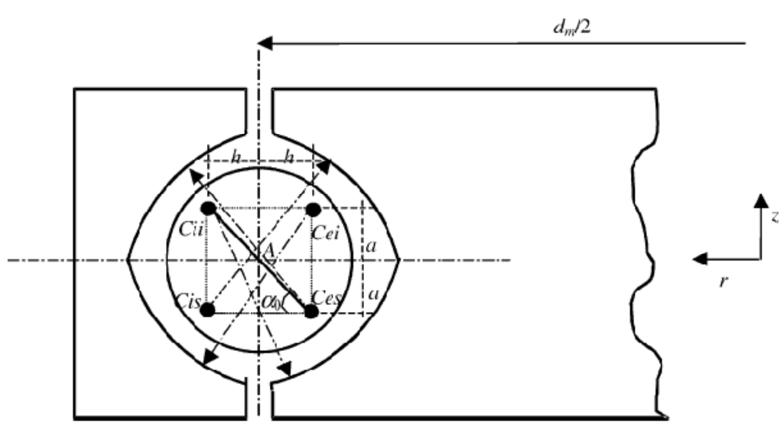


Figure 2.4 Slewing bearing section [4]

• *Cii* represents the centre of curvature of the lower inner raceway.

• *Cis* represents the centre of curvature of the upper inner raceway.

• *Cei* represents the centre of curvature of the lower outer raceway.

• *Ces* represents the centre of curvature of the upper outer raceway.

• *a* and *h* are variables given by the design parameters, such as α_0, ball bearing diameter and conformity (radius of raceway curvature divided bydiameter of ball bearing).

The relative distance between the raceways determines the state of the deformations to which each of the ball bearings is subject, as its angular position (ψ) along the whole of the circumference is known and is shown in Figure 2.5.

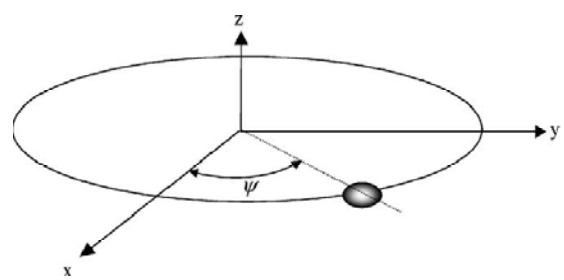


Figure 2.5 Ball bearing's position inside the bearing [4]

The load carried by each ball bearing in the direction of the contact is

$$Q = F(\mathrm{K}, \delta\mathrm{z}, \delta\mathrm{r}, \theta, \mathrm{A}, \alpha_0, \psi) \quad (2)$$

K = Rigidity of the ball bearing/raceway contact

α_0 = Initial contact angle between raceways

ψ = Angle that determines the ball bearing's position inside the bearing

The relation between the deformation of each ball bearing and the force which produces it is given by the contact rigidity, which is a non-linear function of the material of the ball bearing and the raceway and of the relative displacement between them.

$$Q = K\delta^n \tag{3}$$

K = Rigidity of the ball bearing/raceway contact (yield strength of material)

δ = Relative approach between remote points of the bodies in contact

n = load deflection exponent (1.5 for ball bearings)

The forces supported by each ball bearing and the overall displacements of the bearing are related. Lastly, the sum of the projections of the loads of each ball bearing in the three directions gives us the expressions linking the displacements of the bearing and the outer loads. These can be characterised by the following expressions:

$$F_z - \mathrm{KA^n} \sum_0^{2\pi} f_1\,(\alpha_0, \delta z, \delta r, \theta, \psi) = 0 \tag{4}$$

$$F_r - \mathrm{KA^n} \sum_0^{2\pi} f_2\,(\alpha_0, \delta z, \delta r, \theta, \psi) = 0 \tag{5}$$

$$M - \frac{1}{2}\mathrm{d_m KA^n} \sum_0^{2\pi} f_3\,(\alpha_0, \delta z, \delta r, \theta, \psi) = 0 \tag{6}$$

These three equations constitute a non-linear system

2.3.3 Calculation Procedure

The calculation procedure consist of:

i. Defining the initial and final coordinates of the centres of curvature
ii. Studying the contacts between diagonally opposed centres of curvature
iii. Defining the non-linear equations

i. Initial Coordinates of the centres of curvature

Assuming that the radii of curvature of the raceways are equal, the initial coordinates (represented by 1) of the four centres of curvature, without taking the axial clearance into account are:

$$XCii_1 = \left(\frac{d_m}{2} + h\right)\cos\psi \tag{7}$$

$$YCii_1 = \left(\frac{d_m}{2} + h\right)\sin\psi \tag{8}$$

$$ZCii_1 = a \tag{9}$$

$$XCis_1 = \left(\frac{d_m}{2} + h\right)\cos\psi \tag{10}$$

$$YCis_1 = \left(\frac{d_m}{2} + h\right)\sin\psi \tag{11}$$

$$ZCis_1 = -a \tag{12}$$

$$XCei_1 = \left(\frac{d_m}{2} - h\right)\cos\psi \tag{13}$$

$$YCei_1 = \left(\frac{d_m}{2} - h\right)\sin\psi \tag{14}$$

$$ZCei_1 = a \tag{15}$$

$$XCes_1 = \left(\frac{d_m}{2} - h\right)\cos\psi \tag{16}$$

$$YCes_1 = \left(\frac{d_m}{2} - h\right)\sin\psi \tag{17}$$

$$ZCes_1 = -a \tag{18}$$

The initial distance (Figure 2.4) between diagonally opposed centres of curvature is equal to:

$$A = 2\sqrt{(a^2 + h^2)} \tag{19}$$

However, the existing clearance modifies some of the coordinates of the centres of curvature of the Z axis.

If P_r is defined as being the existing radial clearance, r_c as the radius of curvature and d as the diameter of the ball bearing, we have the following relation (Figure 2.6):

$$2(r_c - P_r) - A = d \tag{20}$$

P_r = existing radial clearance

r_c = radius of curvature

d = diameter of ball bearing

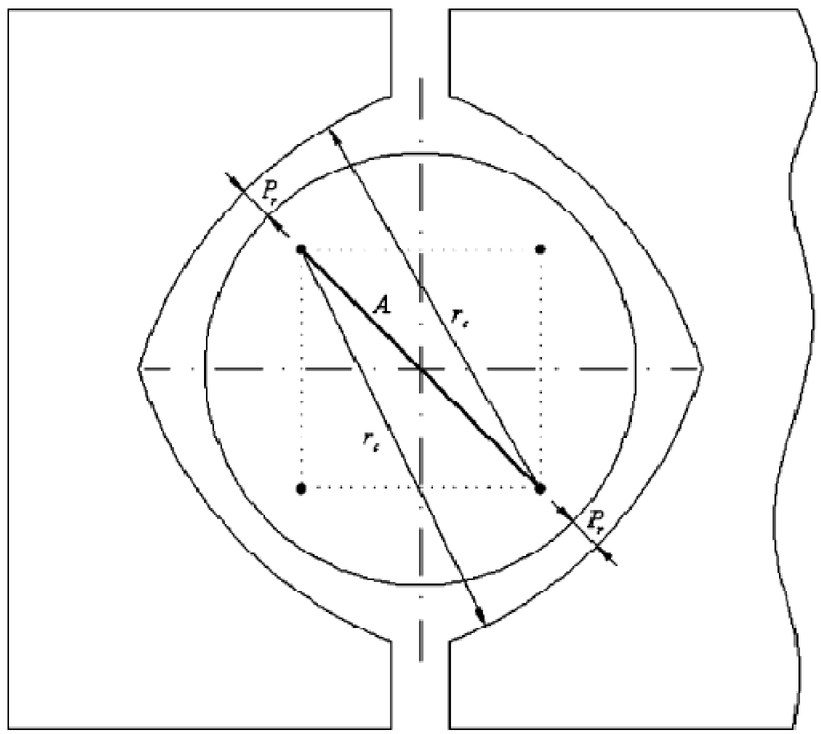


Figure 2.6 Clearance between races and ball bearing [4]

If the inner ring was taken as a reference, the outer ring will move down by an amount *j*, equal to the axial clearance, until contact exists between the sphere and the raceways.

Figure 2.7 shows the positions of the centres of curvature in the initial position, taking into account the axial clearance *j*. The following relationship may be deduced from Figure 2.7:

$$A'^2 = (2h)^2 + (2\text{a} + \text{j})^2 \tag{21}$$

From expression (20), as in the contact $P_r = 0$, it can be deduced that

$$2r_c - A' = d \tag{22}$$

From the expressions (20) and (22) it can be deduced that

$$A' = 2P_r + A \tag{23}$$

Finding j from equation (21) and substituting equation (23) in this equation, the following equation is obtained for the axial clearance:

$$j = \sqrt{(2P_r + A)^2 - (2h)^2} - 2a \tag{24}$$

Therefore, taking the axial clearance into account, the initial coordinates on the z-axis of the centres of curvature *Cei1* and *Ces1* are modified, where

$$ZCei_1 = a - j \tag{25}$$

$$ZCes_1 = -a - j \tag{26}$$

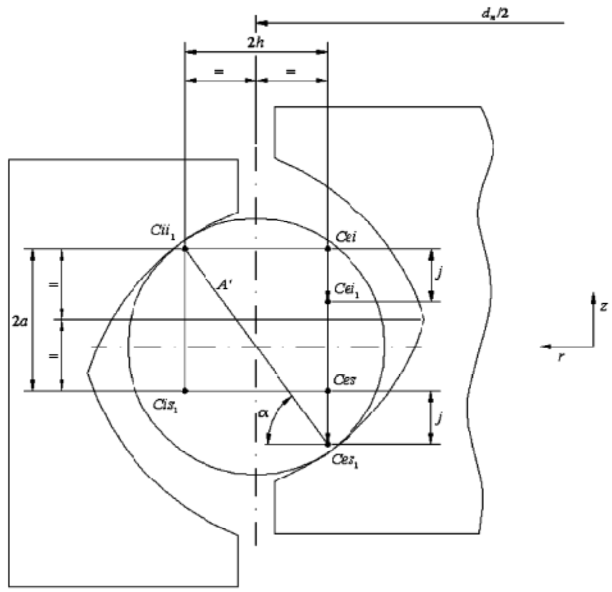


Figure 2.7 Position of the ball when the clearance is absorbed [4]

ii. Final coordinates of the centre of curvature

Once the clearance has been overcome, the outer loads (F_r, F_z and M) are applied on the outer ring, and these cause displacements of the centres of curvature of the outer raceways, δ_r, δ_z and θ.

The moment axis has been established as axis y, and the angle between the radial force and this moment is defined as Φ as can be seen in Figure 2.8.

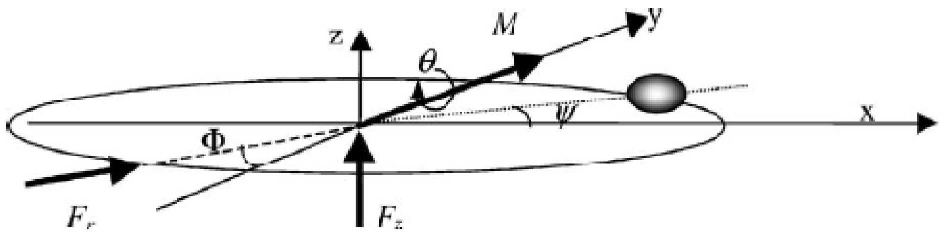


Figure 2.8 Directions of the loads [4]

The final coordinates (represented by 2) of the centres of curvature will be as follows:

$$XCii_2 = XCii_1 \tag{27}$$

$$YCii_2 = YCii_1 \tag{28}$$

$$ZCii_2 = ZCii_1 \tag{29}$$

$$XCis_2 = XCis_1 \tag{30}$$

$$YCis_2 = YCis_1 \tag{31}$$

$$ZCis_2 = ZCis_1 \tag{32}$$

$$XCei_2 = XCei_1 + \delta_r \sin\Phi \tag{33}$$

$$YCei_2 = YCei_1 + \delta_r \cos\Phi \tag{34}$$

$$ZCei_2 = ZCei_1 + \delta_z - \theta(\frac{d_m}{2} - h)\cos\psi \tag{35}$$

$$XCes_2 = XCes_1 + \delta_r \sin\Phi \tag{36}$$

$$YCes_2 = YCes_1 + \delta_r \cos\Phi \tag{37}$$

$$ZCes_2 = ZCes_1 + \delta_z - \theta(\frac{d_m}{2} - h)\cos\psi \tag{38}$$

In a four contact-point bearing with one row of ball bearings, the contact can be happened between two diagonally opposed centres of curvature; the contact can therefore be happened between the centres of curvature *Cii* and *Ces* and/or the centres of curvature *Cis* and *Cei* (Figure 2.9, where in order for the contact to be easily seen the raceways are represented by straight lines) [4].

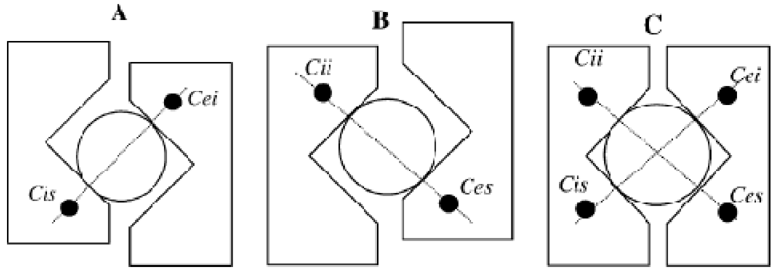


Figure 2.9 Contact between diagonally opposed centres of curvature. [4]

iii. Contact between centres of curvature *Cii* and *Ces*

When the contact is between *Cii* and *Ces* (Figure 2.9A), the distance between the centres of curvature will be

$$A_1 = \sqrt{(XCii_1 - XCes_2)^2 + (YCii_1 - YCes_2)^2 + (ZCii_1 - ZCes_2)^2} \quad (39)$$

The relative displacement between the two centres of curvature will be equal to

$$\Delta_1 = A_1 - A' \quad (40)$$

The contact angle will be

$$\alpha_1 = \arcsin \frac{(ZCii_1 - ZCes_2)}{A_1} \quad (41)$$

The reaction in the ball bearing will be

$$q = K\Delta_1^{\mathrm{n}} \quad (42)$$

Relationship of angles in the contact can be observed in Figure 2.10

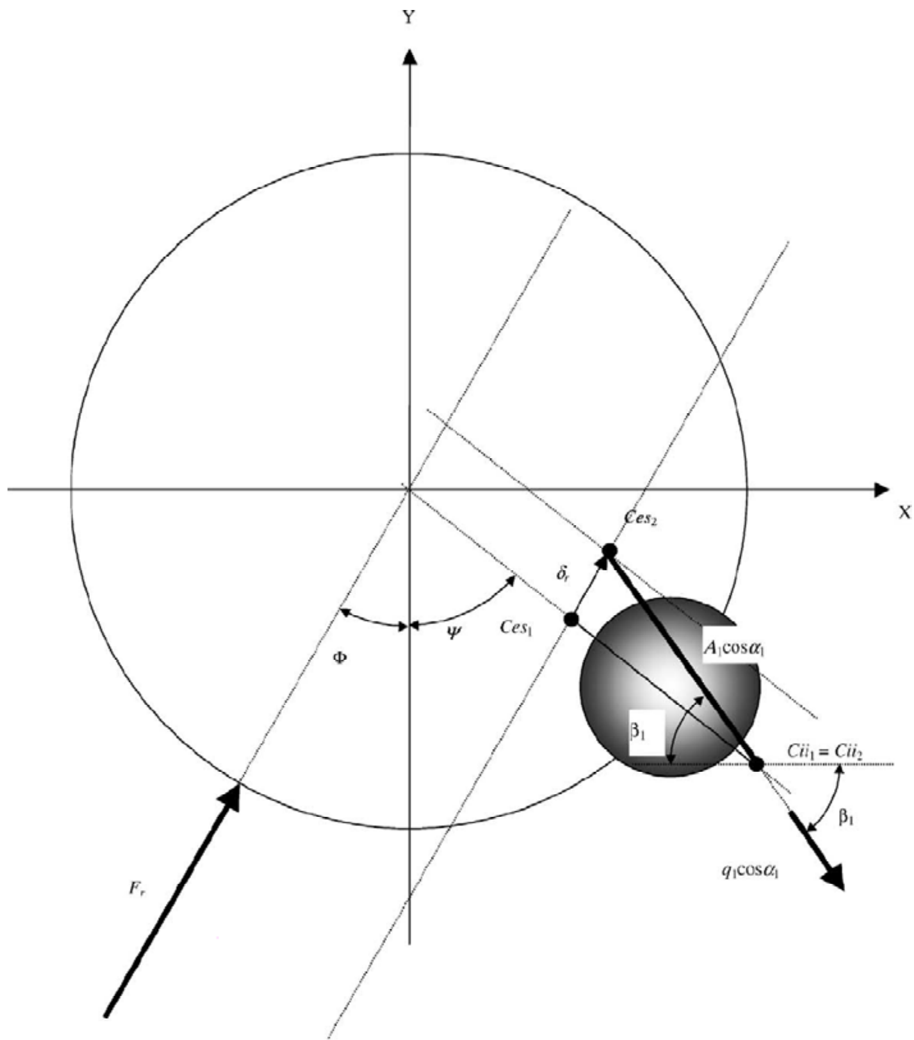


Figure 2.10 Projection of distances, angles and forces on the XY plane in the contact *Cii–Ces* [4]

$A_1 \cos a_1$ is the projection on the XY plane of the distance between centres of curvature *Cii* and *Ces*

$$\text{Cos}\,\beta_1 = \frac{XCii_2 - XCes_2}{A_1 cos\alpha_1} \tag{43}$$

$$\sin\beta_1 = \frac{YCii_2 - YCes_2}{A_1 cos\alpha_1} \tag{44}$$

The x, y, z components of the reaction will be

$$q_{1_x} = q_1 \cos\alpha_1 \cos\beta_1 \tag{45}$$

$$q_{1_y} = q_1 \cos\alpha_1 \sin\beta_1 \tag{46}$$

$$q_{1_z} = q_1 \sin\alpha_1 \tag{47}$$

Substituting (43) in (45) and (44) in (46), we obtain the following:

$$q_{1_x} = q_1 \frac{XCii_2 - XCes_2}{A_1} \tag{48}$$

$$q_{1_y} = q_1 \frac{YCii_2 - YCes_2}{A_1} \tag{49}$$

The vector position of the point of application of the reaction shows the following x, y, z components:

$$R_{1_x} = XCii_2 + \frac{(d-A_1)}{2} \cos\alpha_1 \cos\beta_1 \tag{50}$$

$$R_{1_y} = YCii_2 + \frac{(d-A_1)}{2} \cos\alpha_1 \sin\beta_1 \tag{51}$$

$$R_{1_z} = ZCii_2 + \frac{(d-A_1)}{2} \sin\alpha_1 \tag{52}$$

Where d = diameter of the ball bearing.

Substituting (43) in (50) and (44) in (51), we obtain the following:

$$R_{1_x} = XCii_2 + \frac{(d-A_1)}{2} \frac{XCii_2 - XCes_2}{A_1} \tag{53}$$

$$R_{1_y} = YCii_2 + \frac{(d-A_1)}{2} \frac{YCii_2 - YCes_2}{A_1} \tag{54}$$

The moment of the reaction with respect to the point of origin is

$$M_1 = q_1 * R_1 \tag{55}$$

The components of this moment are

$$m_{1_x} = q_{1_y} R_{1_z} - q_{1_z} R_{1_y} \tag{56}$$

$$m_{1_y} = q_{1_z} R_{1_x} - q_{1_x} R_{1_z} \tag{57}$$

$$m_{1_z} = q_{1_x} R_{1_y} - q_{1_y} R_{1_x} \tag{58}$$

The same procedures are followed in the contact between *Cis* and *Cei* (Figure 2.9B)

iv. Sum of the reactions

Here, all the ball bearings of the slewing bearing are covered and the reactions existing in each ball bearing are added together (if they exist, that is, if the contact works for each of the ball bearings) [4].

The contact for each ball bearing works on the diagonal *Cii–Ces* or *Cis–Cei* or in both cases when $\Delta_1 > 0$ or $\Delta_2 > 0$. When a contact exists on the diagonal of two centres of curvature it is because the centres of curvature move away from each other (if the centres of curvature of the raceways comes closer, the raceways move away):

If $\Delta_1 < 0$, $q_1 = 0$

If $\Delta_2 < 0$, $q_2 = 0$

Thus, δ_r, δ_z and θ must be determined, and must satisfy the following equations:

$$F_r + \sum_{i=0}^{z} \sqrt{q_{1x}^2 + q_{1y}^2} + \sum_{i=0}^{z} \sqrt{q_{2x}^2 + q_{2y}^2} = 0 \quad (59)$$

$$F_z + \sum_{i=0}^{z} q_{1z} + \sum_{i=0}^{z} q_{2z} = 0 \quad (60)$$

$$M + \sum_{i=0}^{z} \sqrt{m_{1x}^2 + m_{1y}^2} + \sum_{i=0}^{z} \sqrt{m_{2x}^2 + m_{2y}^2} = 0 \quad (61)$$

where Z is the total number of the balls in the slewing bearing.

2.4 Selection of Bearing Material

There is a wide range of materials to select from and there is no one ideal bearing material for all cases. The selection depends on the application, which includes tpe of bearing, speed, load, type of lubrication, and operating conditions, such as temperature and maximum contact pressure.[8]

In general, a bearing metal should have balanced mechanical properties. On the one hand, the metal matrix should be soft, with sufficient plasticity to conform to machining and alignment errors as well as to allow any abrasive particles in the lubricant to be embedded in the bearing metal. On the other hand, the metal should have sufficient hardness and compression strength, even at high operating temperature, to avoid any creep and squeezing flow of the metal under load, as well as having adequate resistance to fatigue and impact. The selection is a tradeoff between these contradictory requirements.[8]

2.5 Wear Mechanism

A certain amount of wear is always present. If the sliding materials are compatible, wear can be mild under appropriate conditions such as lubrication and moderate stress. In the case of overloading the bearing or oil starvation, severe wear can develop. The severity of the wear can increase with the surface temperature.

There are four type of wear which are adhesive wear, abrasion wear, fatigue wear and corrosion wear. These types of wear are going to be discuss further.

2.5.1 Adhesive Wear

Adhesive wear can be described as plastic deformation of very small fragments within the surface layer when two surfaces slides against each other [9]. Adhesive wear is associated with adhesion friction, where strong microscopic junctions are formed at the tip of asperities of the sliding surfaces. This wear can be severe in the absence of lubricant. The junctions must break due to relative sliding. The break of a junction can take place not exactly at the original interface, but near it. Small particles of materials are transferred from one surface to another [8].

For two rubbing metals, high adhesion wear is associated with high solid solubility with each other, such as steel on steel [9]. When the temperature at the junction point is relatively high, much stronger junctions are generated. Strong junctions often cause scoring damage. When surface temperature exceeds a certain critical value, wear rate will accelerate [8].

2.5.2 Abrasion Wear

This type of wear occurs in the presence of hard particles, such as sand dust or metal wear debris between the rubbing surfaces. Also, for rough surfaces, plowing of one surface by the hard asperities of the other results in abrasive wear. It is possible to reduce abrasion wear by using soft bearing materials, in which the abrasive particles become embedded, protect the shaft as well as the bearing from abrasion [8].

2.5.3 Fatigue Wear

The damage to the bearing surface often results from fatigue. This wear is in the form of pitting, which can be identified by many shallow pits. The maximum shear stress is below the surface. This often results in fatigue cracks and eventually causes peeling of the surface material [8].

The surfaces in journal bearings are conformal, and the compression stresses are more evenly distributed over a relatively large area. Therefore the maximum compression stress is not as high as in rolling contacts, and adhesive wear is the dominant wear mechanism.

In line and point contact, the surfaces, are not conformal, and fatigue plays an important role in the wear mechanism, causing pitting[8].

2.5.4 Corrosion Wear

Corrosion wear is due to chemical attack on the surface, such as in the presence of acids or water in the lubricant. In particular, a combination of corrosion and fatigue can often cause an early failure of the bearing [8].

2.6 Application of Slewing Bearing

A slewing bearing arrangement consists of a single bearing that can accommodate axial and radial loads as well as tilting moments acting either singly or in combination and in any direction. To fully utilize these bearings, each of the following design considerations must be met [10]:

- The bearing rings must be fully supported around their complete circumference and across the entire width of the axial side faces by strong and rigid associated components (Figure 2.11)
- Strength grade 10.9 attachment bolts are used (EN ISO 898)
- The bearing is properly sealed

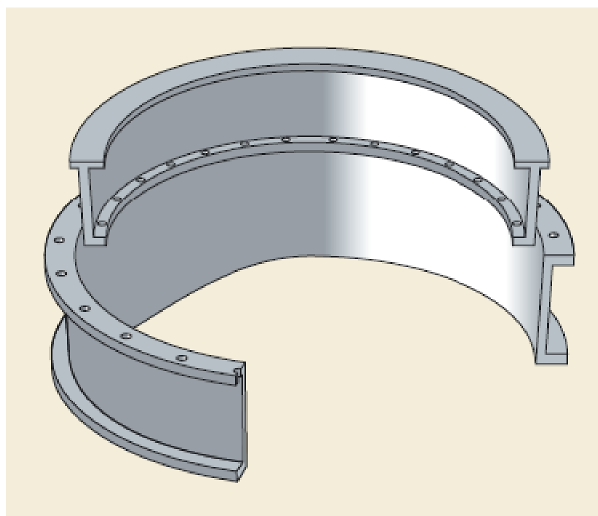


Figure 2.11 Slewing bearing support [10]

2.6.1 Support Structure

Support structures are typically welded frames or castings. Thick-walled cylindrical structures with an inside or outside flange provide better results than thin-walled fabricated structures with a trussed frame (Figure 2.12). Moreover, the arrangement of the walls of the sub- and superstructure should correspond with the rolling element assembly, to optimize power transmission.

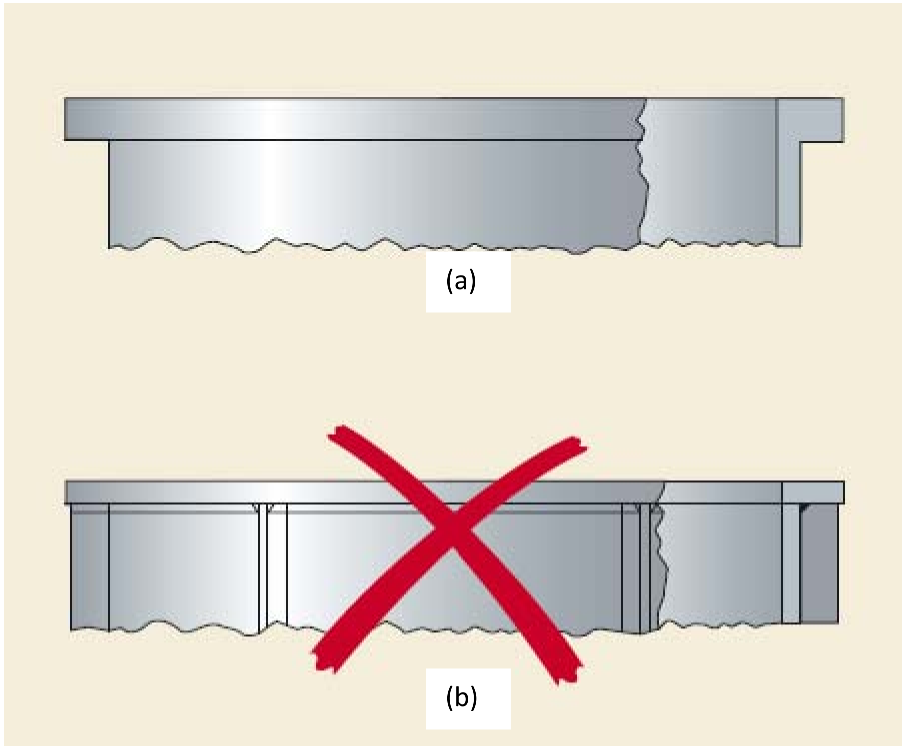


Figure 2.12 (a) Thick-walled cylindrical structures with an inside or outside flange (b) thin-walled fabricated structures with a trussed frame [10]

The flange must support the bearing ring across its entire side face. The thickness of the support flange (Figure 2.13) should be in accordance with the following guideline values [10]:

- $S \geq 0{,}05 \times d_m$, for bearings with a mean raceway diameter ≤ 500 mm
- $S \geq 0{,}04 \times d_m$, for bearings with a mean raceway diameter > 500 mm and $\leq 1\ 000$ mm
- $S \geq 0{,}03 \times d_m$, for bearings with a mean raceway diameter $> 1\ 000$ mm

The requisite minimum wall thickness (Figure 2.13) of the structure can be estimated using;

$S_1 = 0{,}35 \times S$

where

S = thickness of the support flange, mm

S_1 = wall thickness of the structure, mm

d_m = mean raceway diameter of the bearing, mm

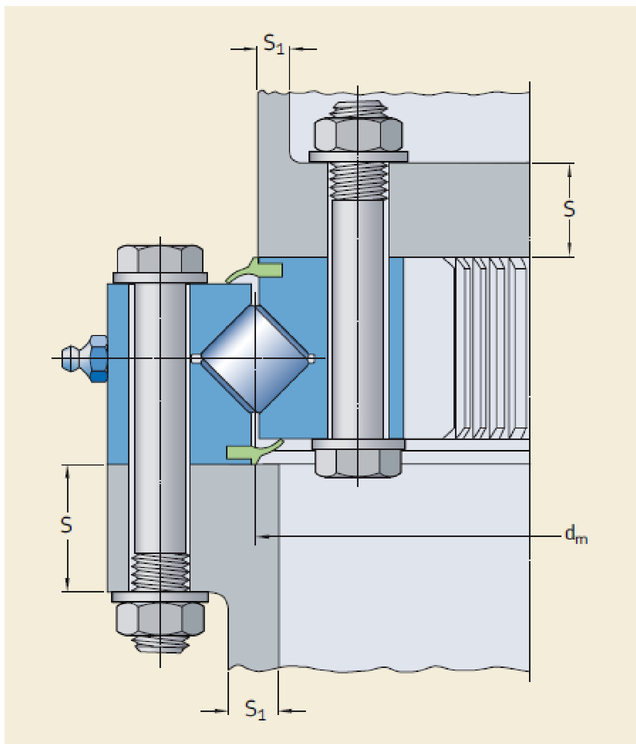


Figure 2.13 Support flange and wall thickness[10]

2.6.2 Support Surfaces

Slewing bearings have limited rigidity, due to their relatively small cross sectional height compared to their diameter. The support structure should therefore be designed for maximum axial and radial rigidity. The support surfaces must be flat and free from rust, paint or burrs. Machining is mandatory and the surface roughness should be within the limits R_a = 3.2 to 6.3 mm. Additionally, the support surfaces should be thoroughly washed and dried before mounting to provide the proper frictional joint between the support surface and the bearing surface [10]. Also, always be sure that the support surfaces are not covered with a preservative or coated with oil or grease.

Before bolting each bearing ring to its support surface, it is essential to check the total axial run-out and flatness of the machined support surfaces, since a low section slewing bearing will be distorted by any irregularities. The flatness should be compared to that of an ideal plane surface. The deviations in height between measuring points of the actual surface drawn over the ideal plane are illustrated in Figure 2.14, the following parameters should be checked prior to mounting.

- Overall flatness tolerance

  The tolerance of the overall flatness in a circumferential direction of the support surfaces (Figure 4) is limited to

  $t_c = (d_m + 1\ 000)/10\ 000$

  where

  t_c = maximum permissible deviation from flatness, mm

  d_m = mean raceway diameter of the bearing, mm

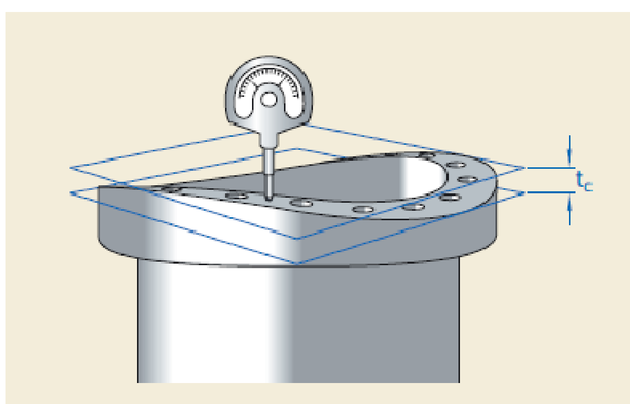


Figure 2.14 Overall flatness tolerance

- Flatness in the circumferential direction

The flatness variation, the difference between the measurements of two consecutive points, as well as the variation in inclination (Figure 2.15), is of great importance. It is measured by dividing the circle into small segments of the length "n" smaller or equal to the distance of the attachment bolt hole. Deviations in the same direction away from the nominal plane, such as measurements "tc" and "t_{cb}", or "t_{cc}" and "t_{cd}" between two consecutive points, should not exceed the guideline value

$$t_{ca} - t_{cb} \leq 0.0002 \text{ x } n$$

If however, the inclination changes direction, as at point P in Figure 2.15, the sum of the deviations, such as measurements "t_{cb}" and "t_{cc}", should not exceed the guideline value

$$t_{cb} + t_{cc} \leq 0.0002 \text{ x } n$$

where,

$t_{ca} - t_{cb}$ = permissible flatness variation between 2 consecutive measurements, where inclination is constant, mm

$t_{cb} + t_{cc}$ = permissible flatness variation between 2 consecutive measurements, where inclination changes direction, mm

n = distance between two consecutive measuring points, mm

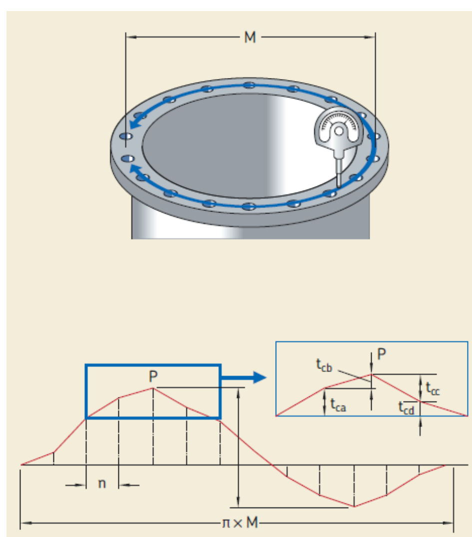


Figure 2.15 Flatness in the circumferential direction

- Flatness in the radial direction
  Flatness in the radial (transverse) direction, e.g. the conicity (Figure 2.16), measured across the width of the support surface is limited to

  $t_t = B / 1000$

  where

  t_t = permissible deviation of axial run-out in the radial (transverse) direction, mm

  B = width of the support surface, mm

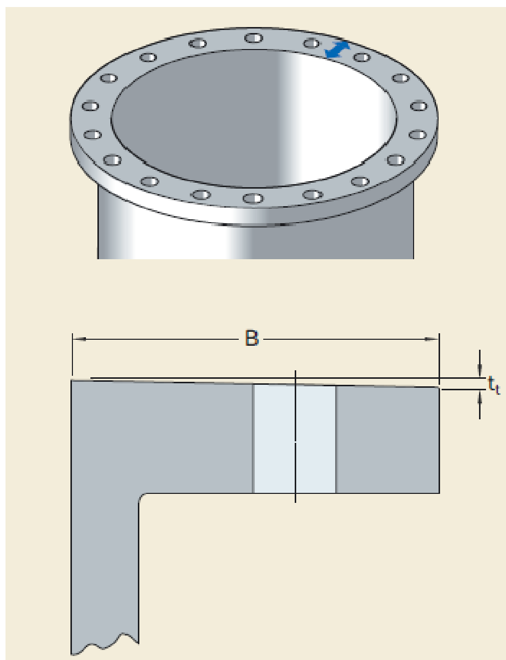


Figure 2.16 Flatness in the radial direction [10]

CHAPTER 3

METHODOLOGY

Research on the concept and design of slewing bearing are to be done. Then, a real life problem will be anticipated and calculated. Based on the problem anticipated, the bearing design calculation on the distribution of load will be done. After that, the bearing will be designed using CATIA. After finalization of the design, fabrication process can be started. Testing process will take place after the fabrication phase is completed. Then, proper documentation of the whole process will be completed.

3.1 Flow Chart

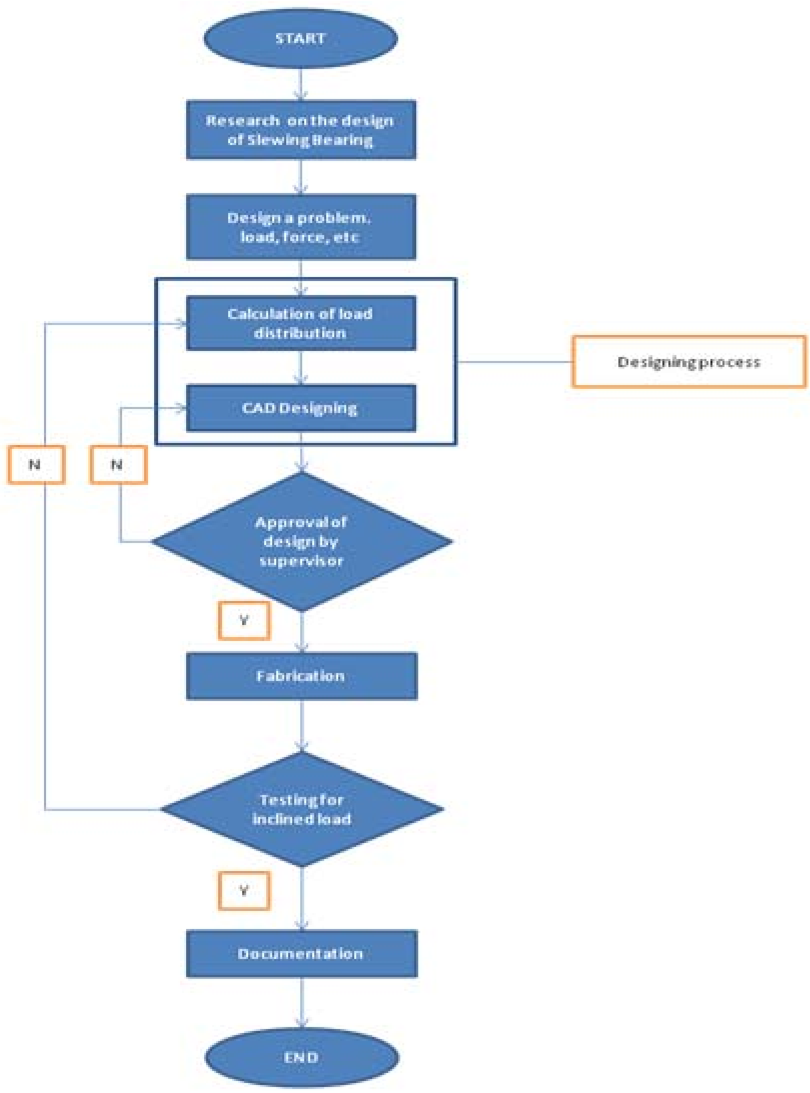


Figure 3.1 Flow diagrams of activities such as designing, fabrication and testing

3.2 Tools and Equipment Required

During the progress of the project, some equipment and tools are needed in the design and fabrication of the prototype of innovative solar water heater. The tools and equipment uses are:

Software:

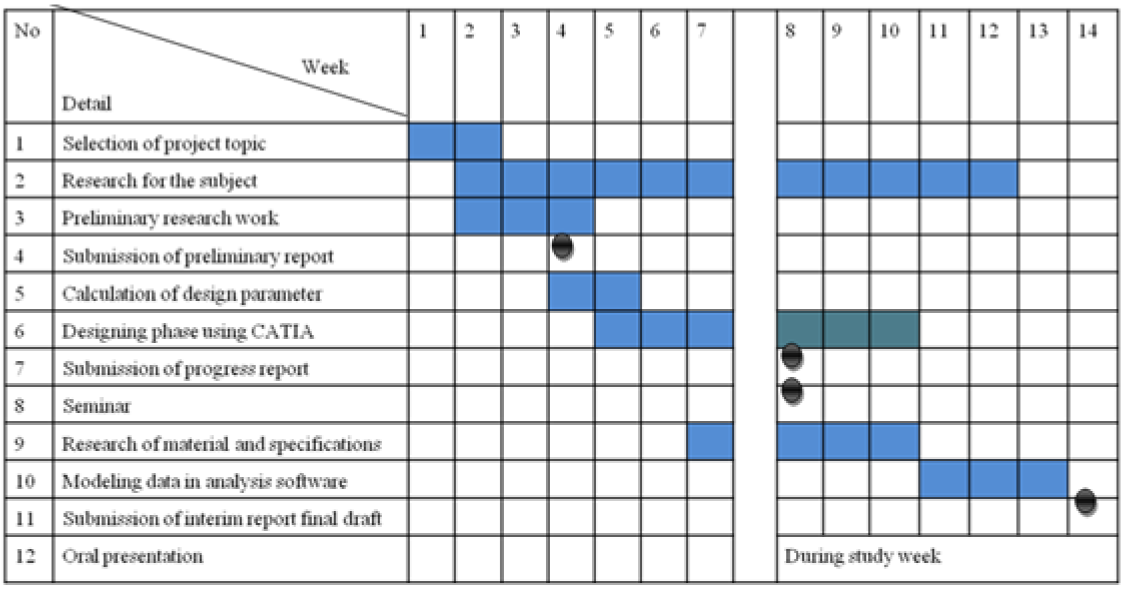
- CATIA
- Auto-CAD
- ANSYS
- Microsoft Office

Tools & Equipments

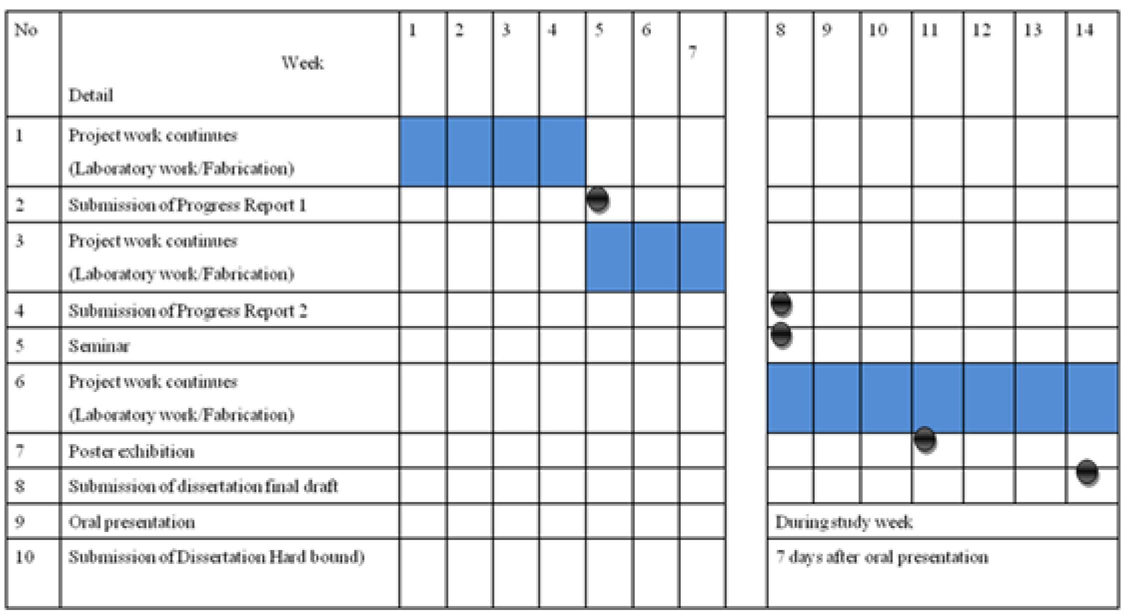
- Welding Machine (SMAW)
- Drilling machine
- CNC lathe and milling machine
- Fabrication tools

3.3 Gantt Chart

FYP 1 Gantt Chart



FYP 2 Gantt Chart



3.4 Designing Method

1. Using formula in (2.3.3 Calculation Procedure, pg 8) the dimension for the slewing bearing, the initial coordinates for raceways centre of curvature, and also the maximum force it can sustain was obtained. The equations were included in Microsoft Excel for ease of calculations. Calculations are made with regard to several assumptions and also with trial and error method. For the calculations, several assumptions were made:
   i. Bearing diameter, d_m was assumed as 200mm, based on the size of application.
   ii. Radius of curvature, r_c was assumed to be 10.75mm and this will directly affect the diameter of steel ball which is 12mm.
   iii. Angle between the radial force and moment, θ was assumed to be between 20^o - 45^o because the force exerted was based on slow moving application.
   iv. Variables *a* and *h* were both assumed 3mm (ratio 1:1).

2. As for the material for the bearing, Alloy Steel 1040 was used. The yield strength value of AISI 1040 which is 353.4 Mpa was used as the value of non-linear function of the material of the ball bearing and the raceway, *K* in calculating the contact rigidity of the slewing bearing. The chemical composition of the material was listed in Table 3.1 below:

Element	Weight %
C	0.37-0.44
Mn	0.60-0.90
P	0.04 (max)
S	0.05 (max)

Table 3.1 Properties of AISI 1040 [7]

The mechanical properties of AISI 1040 are shown below in Table 4.2:

Properties		Conditions	
		T (°C)	Treatment
Density (×1000 kg/m^3)	7.845	25	
Poisson's Ratio	0.27-0.30	25	
Elastic Modulus (GPa)	190-210	25	
Tensile Strength (Mpa)	518.8	25	annealed at 790°C more
Yield Strength (Mpa)	353.4		
Elongation (%)	30.2		
Reduction in Area (%)	57.2		
Hardness (HB)	149	25	annealed at 790°C more
Impact Strength (J) (*Izod*)	44.3	25	annealed at 790°C more

Table 3.2 Mechanical properties of AISI 1040 [7]

3. From the results obtained (the dimension and also the initial coordinates for raceways centre of curvature), a drawing was made in CATIA. The steps taken are as follow:
    i. Specify the coordinates for the raceways curvature in CATIA and draw the raceways curvature based on the result of the calculations.

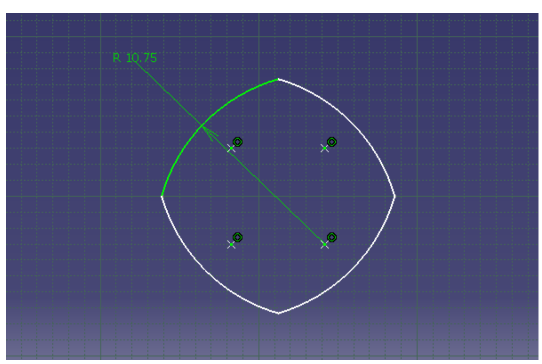


Figure 3.2 The initial coordinates and the raceways curvature in CATIA

ii. Draw the inner and outer ring shape (planar view) of the slewing bearing.

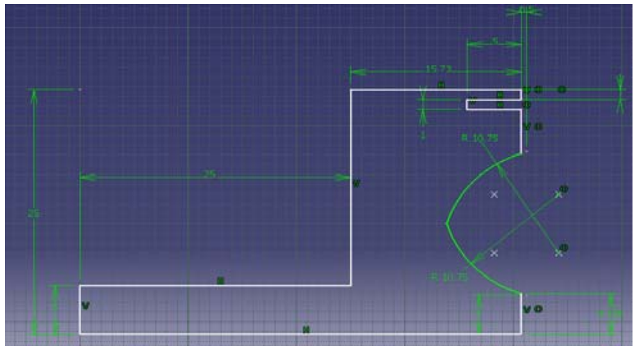


Figure 3.3 Outer ring planar view

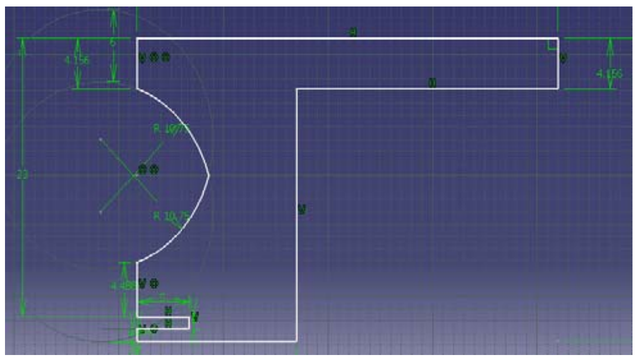


Figure 3.4 Inner ring planar view

iii. Then, the drawing was rotated around its axis to create a 3D shape of the outer and inner ring. Then the outer ring and inner ring were assembled together with steel ball inside.

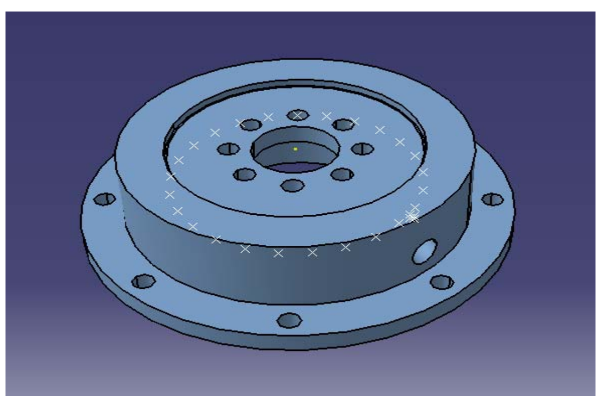


Figure 3.5 Assembled slewing bearing

3.5 Testing Method

The test rig for the Slewing bearing has been design according to the size of the scaled down model. The test rig was design as such to imitate real application. The test rig consists of:

1. Upper body
2. Lower body
3. Weight

The rig is connected to the slewing bearing using M5 bolts and nuts. The Testing Rig will be use to test for load of 10kg and also the axial, radial and rotational displacement of the Slewing Bearing

Figure 3.6 Upper body of Slewing Bearing testing rig

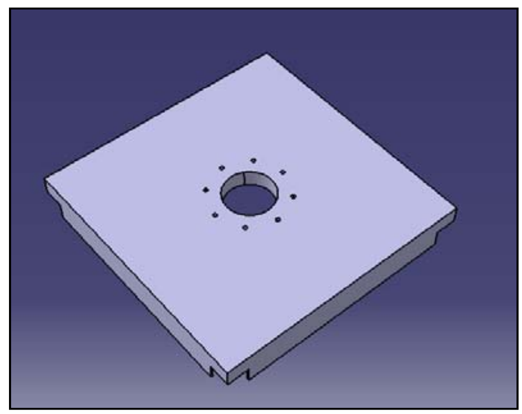


Figure 3.7 Lower body of Slewing Bearing testing rig

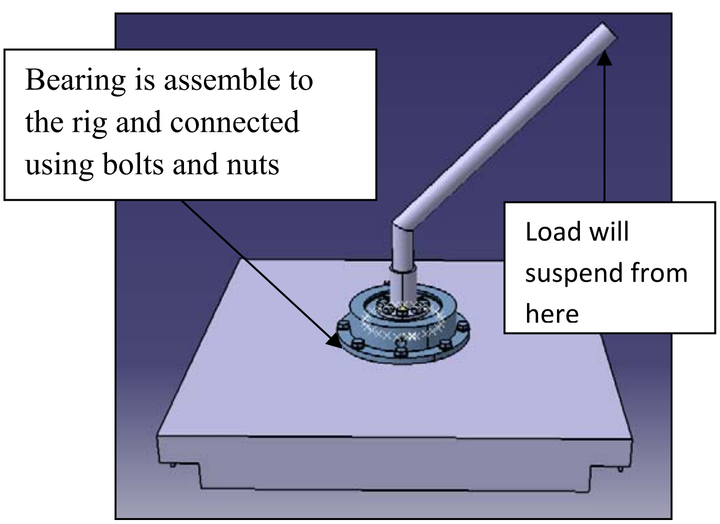
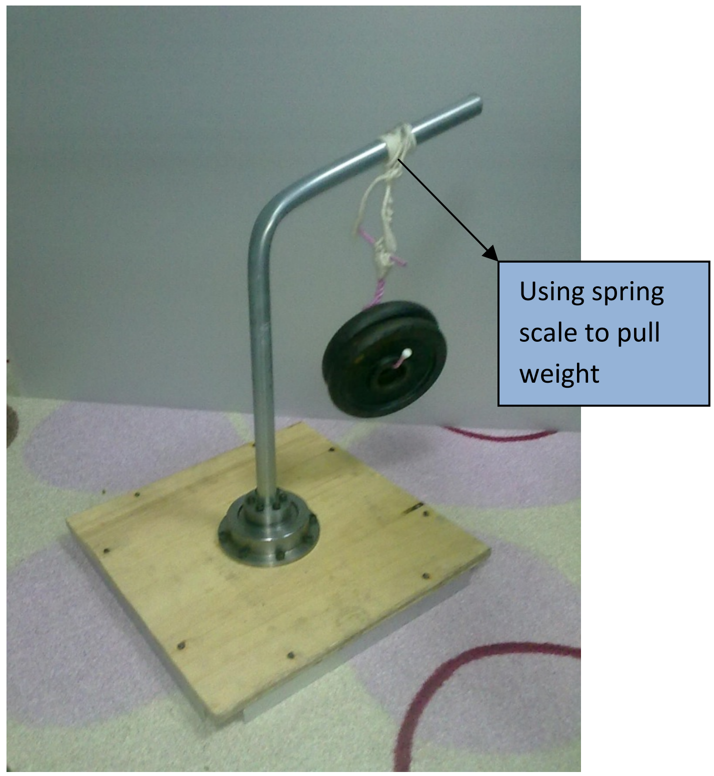


Figure 3.8 Complete assembly of the Slewing Ring to the test rig

Slewing bearing was tested for the smoothness of its movement. Spring scale for measuring force in Newton was used as a measuring device. The spring scale is applied perpendicular to the rod to measure the force needed to move the test rig for different hanging weight. Results obtained were recorded. Picture below shows how the experiment was conducted.



Picture 3.1 Test rig assembly

Picture 3.2 Spring Scale

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Calculation Result

After putting in all the values in the excel spreadsheet, the following result was obtained (Table 4.1) and can be used as design dimensions of the slewing bearing.

Design Parameters	**Value**
Bearing diameter, d_m (mm)	100
Variable, a (mm)	0.5
Variable, h (mm)	0.5
Radius of curvature, r_c (mm)	4.25
Initial distance between diagonally opposed centres of curvature, A (mm)	1.414
Number of ball bearings, Z	25
angular position of the ball bearing within the bearing, ψ (deg)	14.4
Diameter of ball bearing, d (mm)	6.081
Existing radial clearance, P_r (mm)	0.5
Angle between the radial force and moment, θ (deg)	30

Table 4.1 Design parameters of scaled down slewing bearing

The initial and final coordinate of the centre of the radius of curvature also was calculated. The coordinates will be used in modeling the slewing bearing in CATIA. The coordinates are shown in Table 4.2 and Table 4.3 below:

X-axis	Cii	Cis	Cei	Ces
	29.54	29.54	28.57	28.57
Y-axis	Cii	Cis	Cei	Ces
	7.58	7.58	7.33	7.33
Z-axis	Cii	Cis	Cei	Ces
	0.50	-0.50	0.50	-0.50

Table 4.2 Initial coordinate of the centre of the radius of curvature for the scaled down slewing bearing

X-axis	Cii	Cis	Cei	Ces
	29.54	29.54	29.07	29.07
Y-axis	Cii	Cis	Cei	Ces
	7.58	7.58	8.20	8.20
Z-axis	Cii	Cis	Cei	Ces
	0.50	-0.50	-3.98469	-4.98469

Table 4.3 Final coordinate of the centre of the radius of curvature for the scaled down slewing bearing

From the calculation also, the maximum load the slewing bearing can sustain has been determined. The maximum load depends mostly on the number of ball bearings used and the type of material used. The results are as below:

$$F_r = 1.17 \times 10^{10}\ \text{N}$$

$$F_z = 2.35 \times 10^{10}\ \text{N}$$

$$M = 6.96 \times 10^{11}\ \text{N.m}$$

The engineering drawing of the scaled down slewing bearing is attached in Appendix I.

For sealing purposes, O-Ring for rotary shaft application was used to seal the raceways so that the lubrication oil will not come out from the clearance of the outer ring and the inner ring. A gland has been design along the inner circumference of the outer ring for the O-Ring.

4.2 Discussion on the design of the slewing bearing

The design has been improvised in term of practicality and also in term of weight reducibility. The previous design (Figure 4.1) is not practical because there are no ways to put in the ball into the bearing's raceways. The latest design was designed with a hole slightly bigger than the steel ball and oriented 20^{o} from the surface of the outer ring. However, after further discussion with the machinist, I decided to modified the hole which is then oriented 90^{o} for ease of machining process. For better view of the design, refer to Appendix I.

Also, the latest design has been reduced in weight. Some unwanted part of the bearing which does not play any significant role will be machined out to reduce the weight of the bearing. The new design also can sustain more tilting forces compared to the old one. This is because the number of steel balls in the raceways has been added.

Sealing O-ring also has been chosen as a sealant for the Slewing Bearing. This is because it is cheap, easy to assemble and also easy to get. O-ring for rotary applications was chosen because it has more durability for axial stress.

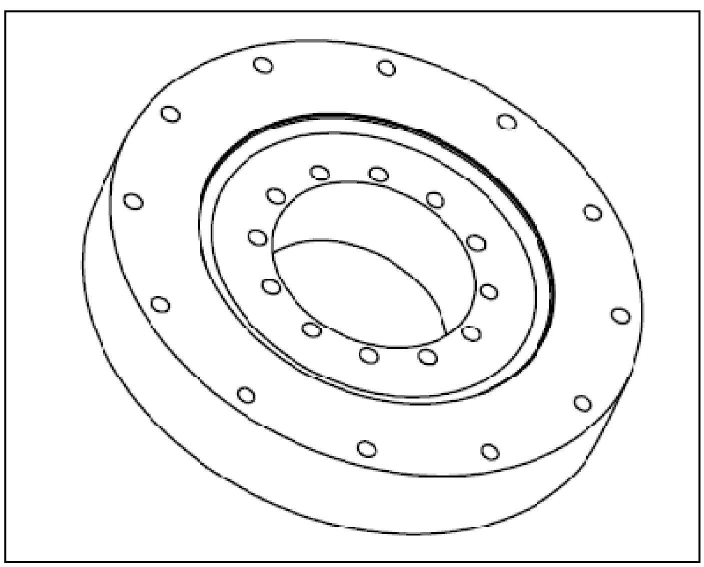


Figure 4.1 Previous slewing bearing design

4.3 Fabricated Slewing Bearing

Below is the finished slewing bearing. The fabrication of slewing bearing was outsourced and the cost is RM 300.00. Some modifications were done due to fabrication difficulty. The modifications are:

1. The hole for inserting steel ball was oriented 90^{o} and the hole was drilled on the outer ring.
2. The hole for bolts was initially eight (8) holes altogether on the inner ring but after realizing that the holes were too close together, it was decided to drill six (6) holes instead of eight (8)

Picture 4.1 Complete assembly of Slewing Bearing

Picture 4.2 Outer ring of Slewing Bearing

Picture 4.3 Inner ring of Slewing Bearing

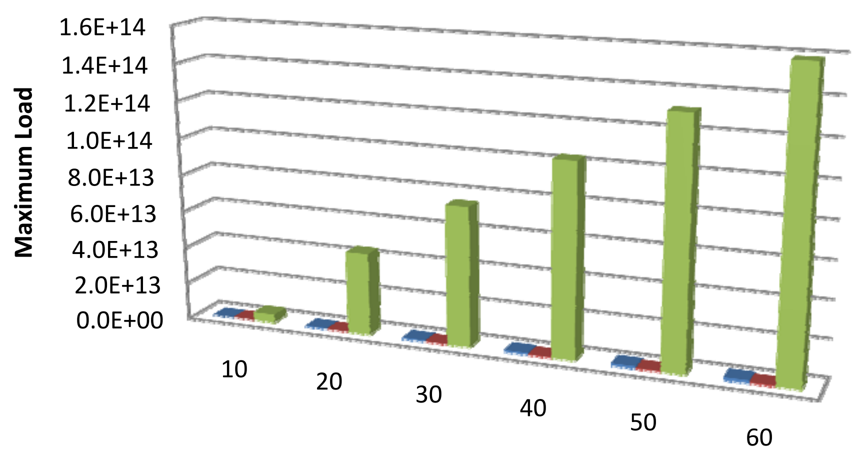
Picture 4.4 Hole plug of Slewing Bearing

Picture 4.5 Steel ball of Slewing Bearing

Picture 4.6 O-ring of Slewing Bearing

4.4 Results from Calculations and Test

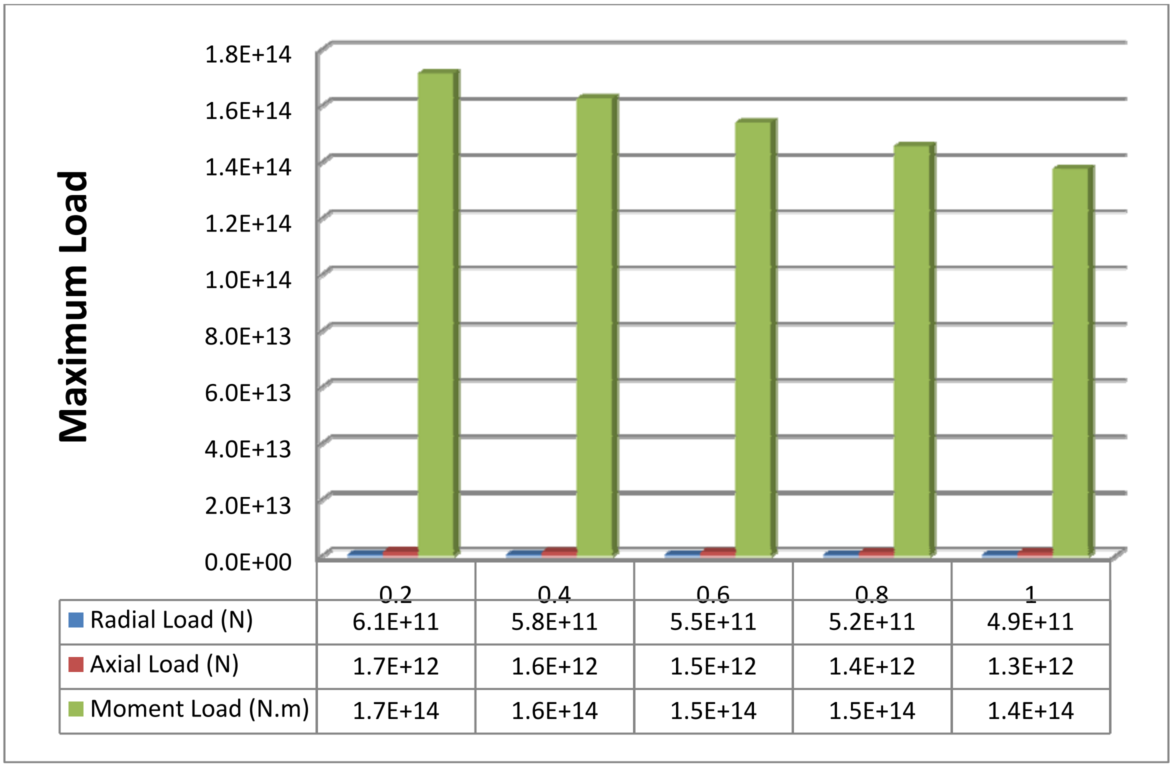
i. Analysis results from calculations



	10	20	30	40	50	60
Axial Load (N)	5.3E+10	4.5E+11	7.4E+11	1.0E+12	1.3E+12	1.5E+12
Radial Load	4.3E+10	1.7E+11	2.7E+11	3.7E+11	4.7E+11	5.7E+11
Moment Load (N.m)	4.4E+12	4.4E+13	7.5E+13	1.0E+14	1.3E+14	1.6E+14

Figure 4.2 Graph showing maximum loads the ball bearing can sustain relative to the number of steel balls.

From the graph, it can be seen that the maximum load the slewing bearing can sustain increases with the number of balls. This means, increasing the number of balls can increase the load rating of the bearing.



	0.2	0.4	0.6	0.8	1
Radial Load (N)	6.1E+11	5.8E+11	5.5E+11	5.2E+11	4.9E+11
Axial Load (N)	1.7E+12	1.6E+12	1.5E+12	1.4E+12	1.3E+12
Moment Load (N.m)	1.7E+14	1.6E+14	1.5E+14	1.5E+14	1.4E+14

Figure 4.3 Graph showing maximum loads the ball bearing can sustain relative to the existing radial clearance.

The maximum load of the slewing bearing can sustain decreases with the increase of the radial clearance. From both Figure 4.2 and Figure 4.3, we can see that the moment load this bearing can sustain is much higher than the radial and axial load. This mean this bearing is suitable to use in inclined load applications. Radial clearance of a slewing bearing are showed in figure 4.4 below.

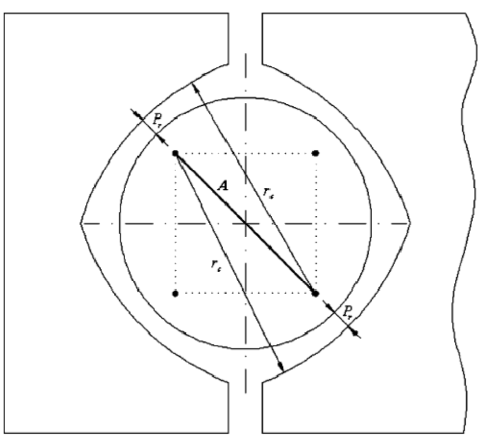


Figure 4.4 Figure shows radial clearance denoted by P_r [4]

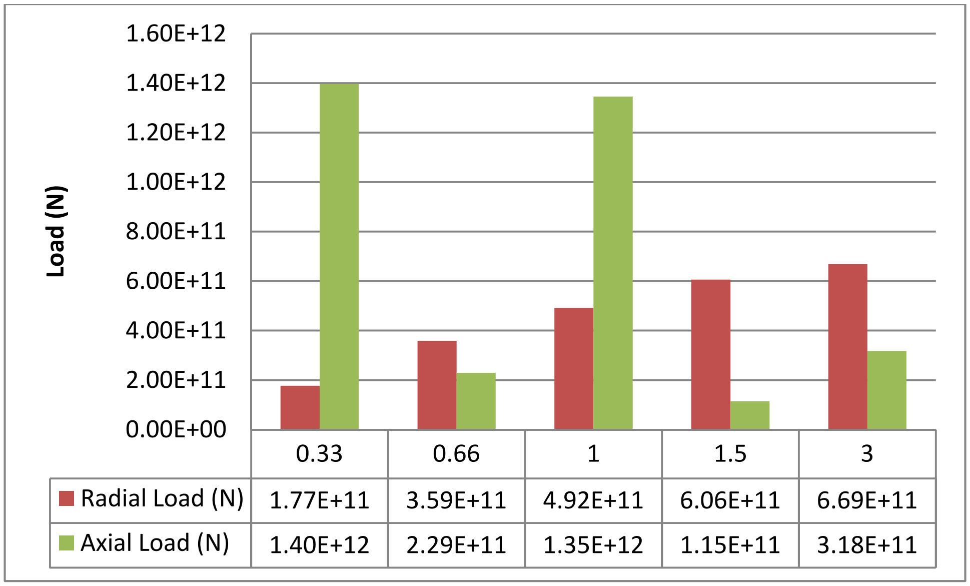


Figure 4.5 Graph showing maximum radial and axial loads the ball bearing can sustain relative to the ratio of h and a

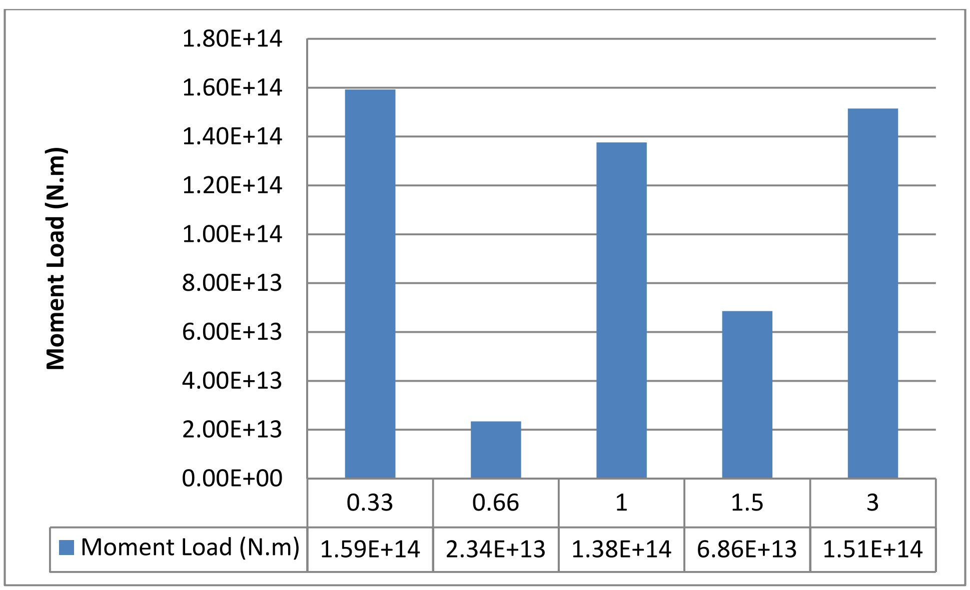


Figure 4.6 Graph showing maximum moment loads the ball bearing can sustain relative to the ratio of h and a

The value of *a* and *h* is the value of the distance between the centers of curvature. As the ratio of *a* and *h* changes, the resulting maximum forces it can sustain also change. The parameter of *a* and *h* are shown in Figure 4.7:

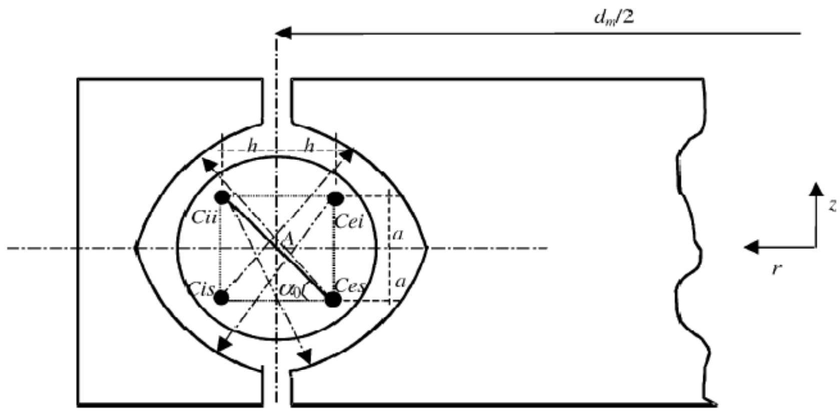


Figure 4.7 Figure shows the parameter *a* and *h*

ii. Analysis results from testing

Figure 4.8 below shows the force needed to pull the weight as shown in methodology of how the experiment was conducted. The purpose is to show that the bearing is functioning in an almost frictionless condition. From Figure 4.8, we can see there are increments in force needed to pull the weight but very little increment and the value of the force in Newton is too small that it can be neglected.

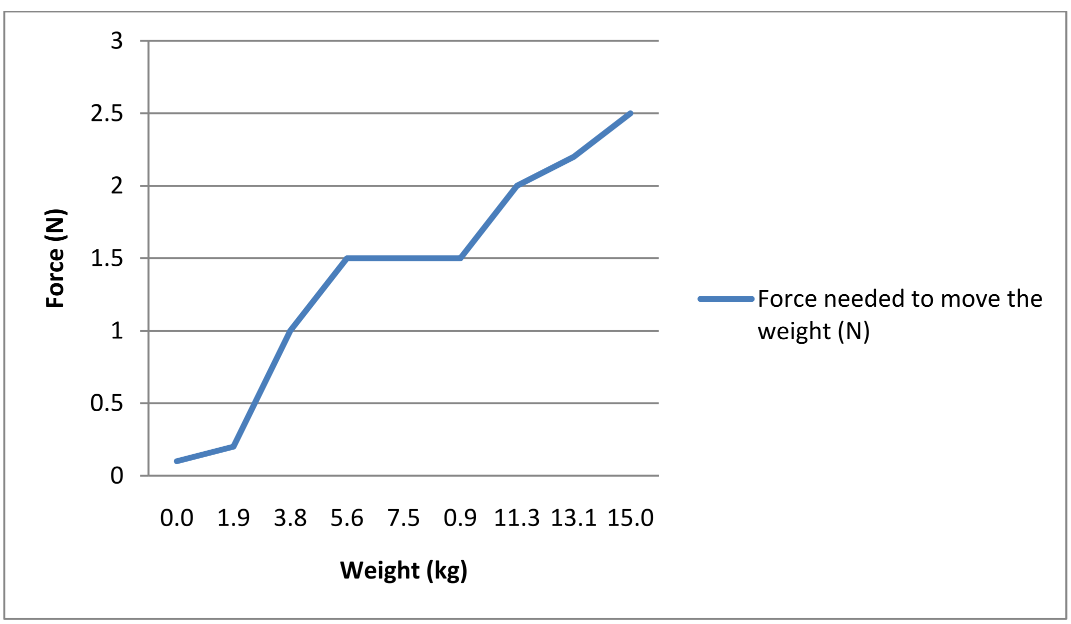


Figure 4.8 Graph showing force needed to pull the weight relative to the mass of the weights.

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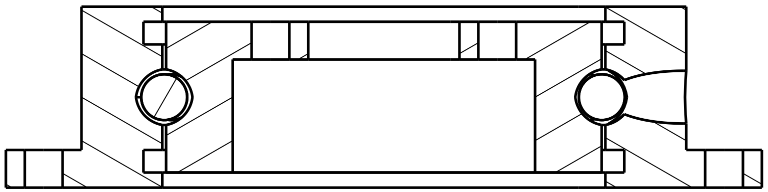
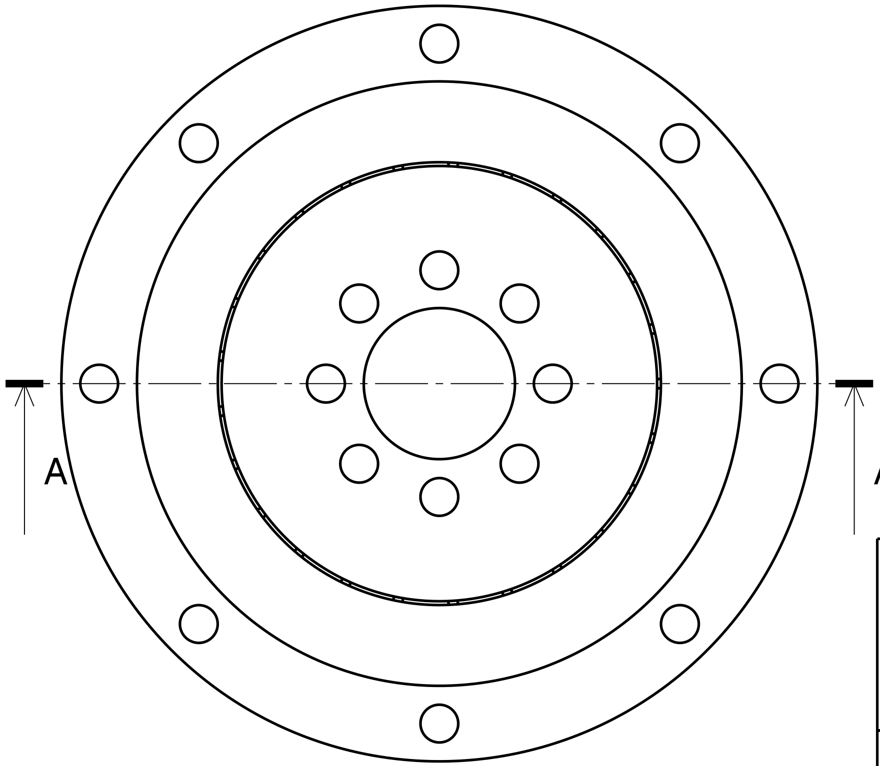
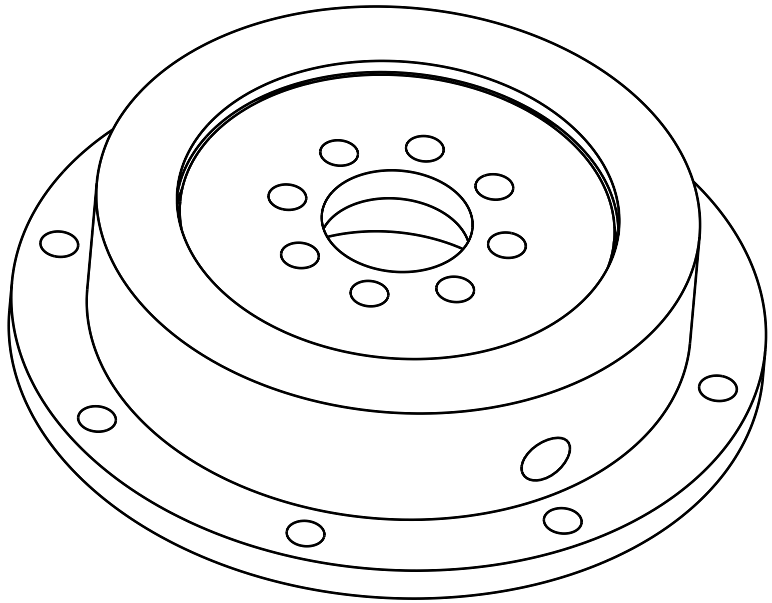
APPENDIX I

Engineering Drawing of Slewing Bearing







GENERAL TOLERANCES	≤3	>3 AND ≤6	>6 AND ≤30	>30 AND ≤120	>120 AND ≤400	>400 AND ≥1000	>1000 AND ≤2000	>2000 AND ≤4000	>4000 AND ≤8000	>8000 AND ≤12000
MACHINING	± 0.1	−0.1	−0.2	−0.3	−0.5	−0.8	−1.2	−2.0	−3.0	−4.0
FABRICATION	-	-	−0.1	−0.1	−0.1	−2.0	−3.0	−4.0	−5.0	−6.0

ALL DIMENSIONS ARE IN 'mm' UNLESS OTHERWISE SPECIFIED

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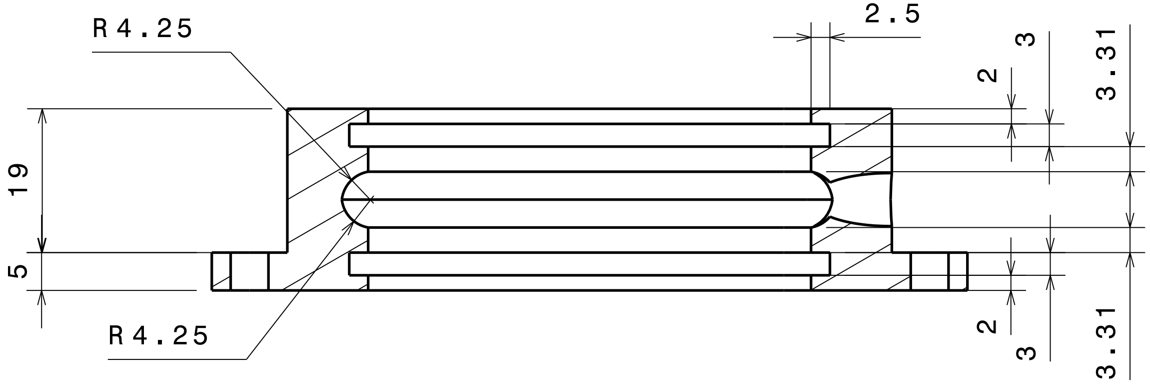
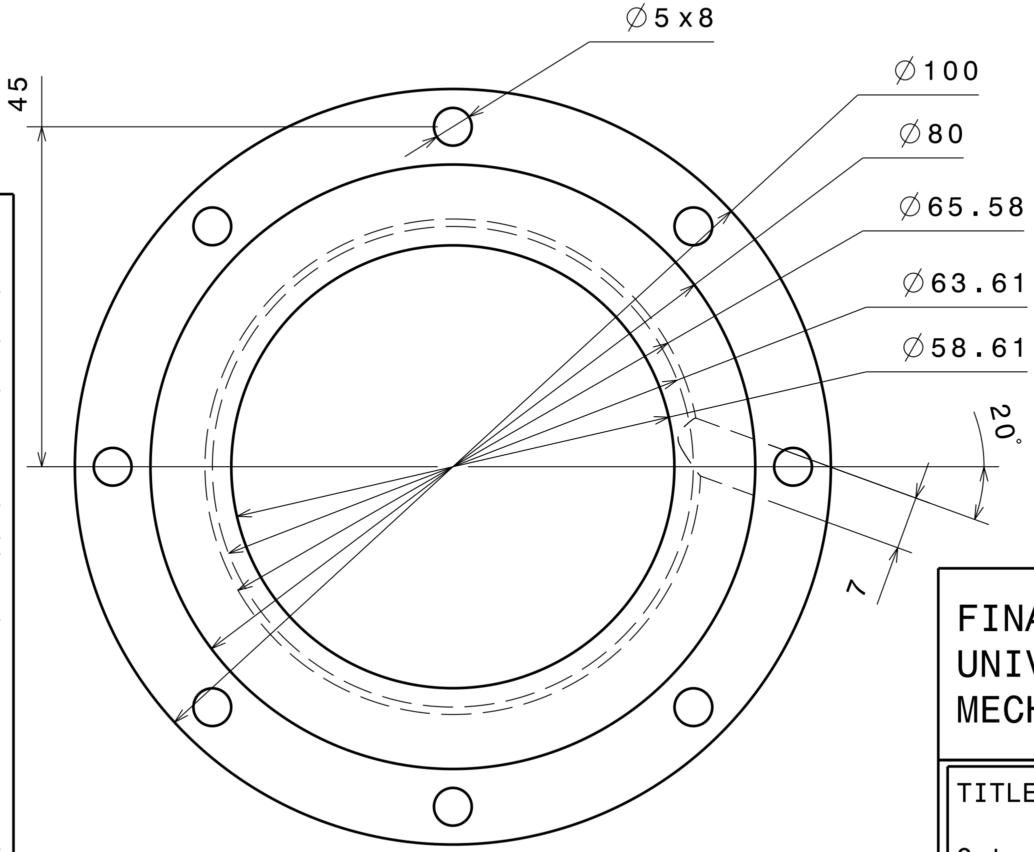
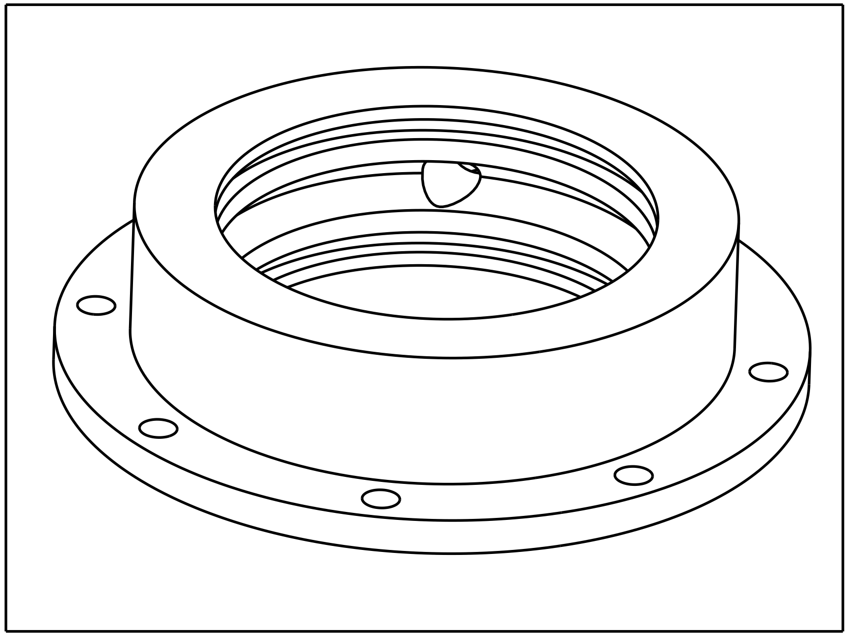
TITLE: Slewing Bearing Assembly	DATE: 21 Aug 2010	FORMAT: A4
	DWG. NO.:UTP-ME-FYP-00 (1)	
DRAWN BY: AHMAD RAFIE B. MOHMAD TAHIR STD. ID: 9000	CHECKED BY: DR. GILANI	







GENERAL TOLERANCES	≤3	>3 AND ≤6	>6 AND ≤30	>30 AND ≤120	>120 AND ≤400	>400 AND ≥1000	>1000 AND ≤2000	>2000 AND ≤4000	>4000 AND ≤8000	>8000 AND ≤12000
MACHINING	± 0.1	−0.1	−0.2	−0.3	−0.5	−0.8	−1.2	−2.0	−3.0	−4.0
FABRICATION	-	-	−0.1	−0.1	−0.1	−2.0	−3.0	−4.0	−5.0	−6.0

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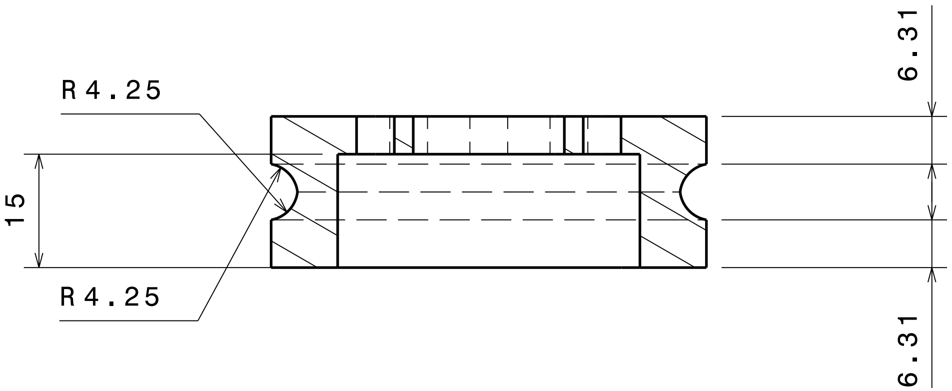
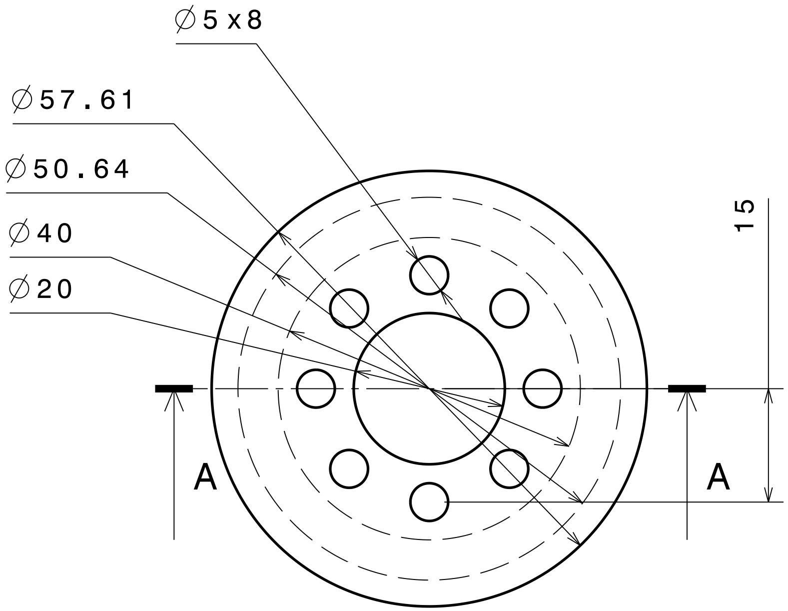
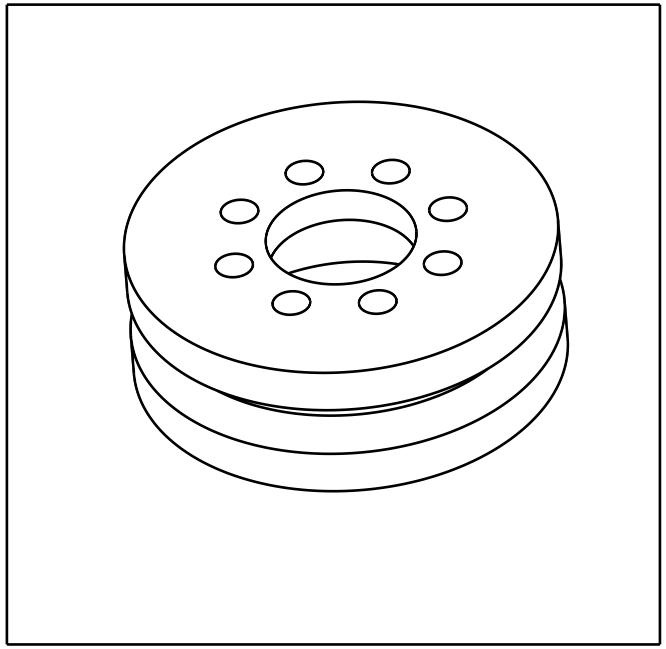
TITLE: Outer Ring	DATE: 21 Aug 2010 / FORMAT: A4
	DWG. NO.: UTP-ME-FYP-01(1)
DRAWN BY: AHMAD RAFIE B. MOHMAD TAHIR STD. ID: 9000	CHECKED BY: DR. GILANI







GENERAL TOLERANCES	≤3	>3 AND ≤6	>6 AND ≤30	>30 AND ≤120	>120 AND ≤400	>400 AND ≥1000	>1000 AND ≤2000	>2000 AND ≤4000	>4000 AND ≤8000	>8000 AND ≤12000
MACHINING	± 0.1	−0.1	−0.2	−0.3	−0.5	−0.8	−1.2	−2.0	−3.0	−4.0
FABRICATION	-	-	−0.1	−0.1	−0.1	−2.0	−3.0	−4.0	−5.0	−6.0

ALL DIMENSIONS ARE IN 'mm' UNLESS OTHERWISE SPECIFIED

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TITLE: Inner Ring	DATE: 21 Aug 2010
	FORMAT: A4
	DWG. NO.: UTP-ME-FYP-02(1)
DRAWN BY: AHMAD RAFIE B. MOHMAD TAHIR STD. ID: 9000	CHECKED BY: DR. GILANI

APPENDIX II

EDX (Engineering Design Exhibition) Certificate

Certificate of Appreciation

This certificate is awarded to

AHMAD RAFIE BIN MOHMAD TAHIR
880521-14-5079

for his/her participation as

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in recognition of your contribution in the twenty-sixth edition of

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